Analysis of a Flat-Plate Solar Collector System for Pulp Drying

A THEORETICAL-IDEAL ANALYSIS BASED ON AN EASTERN MAINE PULP MILL

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Purpose & Location

The purpose of this project is to analyze and configure a solar-thermal heating system for use in a paper mill setting. The location chosen in this case is Woodland, ME. Located there is Woodland Pulp LLC, a pulp mill that has been operating for over 100 years. Additionally, on the property, there is a tissue mill, St. Croix tissue. During my time working there, I became well acquainted with the steam infrastructure and the degree to which it was implemented throughout the mill. It is for these reasons that I selected this location to analyze the viability and performance of a solar-thermal collection and steam production system.

Since this facility dries both wood pulp and tissue paper, the steam load is very high. In this project an ideal solar thermal process, detailed in the figure below, will be analyzed and sized for the components determined in order for the plant to operate 100% on solar power.

Process Plan

Considering the diagram to the right, which depicts the overall system to be designed, this analysis will require numerous steps to complete. As such, having a plan is crucial to a successful and efficient analysis.

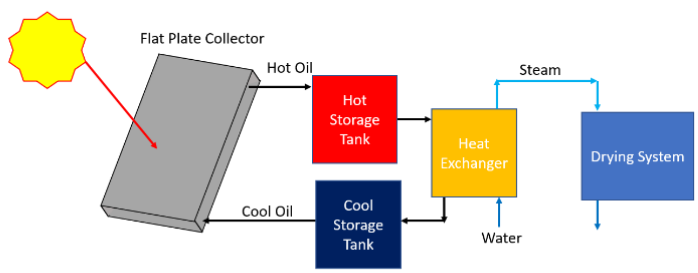


Figure 1: Solar-Thermal Drying System Diagram

Perhaps the most complex part of the analysis will be the flat plate solar collector. Performing an analysis on such a device takes significant time and work in order to be effective, and thus will be done last. That way, as many variables as can be solved for before hand will be known, minimizing the difficulty of sizing the system. The analysis will begin with the load from the drying system. This load will be calculated in a rate form for both first and second shift, so that at any time throughout the year, the minimum system output will be known. From there, the heat exchanger will be analyzed to derive the necessary mass flow rates for different times of day. Such a derivation is possible due to given information about the hot oil temperature and output of the solar collector. Using this data and engineering intuition, design parameters for the storage tanks will be determined. Finally, the solar collector will be analyzed and sized to fit the storage systems and a final analysis will be done to find the optimal system parameters.

Finally, this project will make use of TMY data, or "Typical Meteorological Year", sourced from the United States Geological Survey. This data consists of measurements of direct normal radiation and atmospheric conditions throughout the year.

Energy Load from Dryers

In order to adequately size the solar collector, as well as the storage unit, it is necessary to determine the load drawing from the system. This will be done on a rate basis for use later in deriving heat transfer from the exchanger for mass flow calculation.

In order to find the heat draw by the dryers, the inlet temperature of the water and the temperature of the steam are needed. These would generally be found using sensors installed at the plant, but for this project we will assume them as follows:

$$Water\ Inlet\ Temperature = 15C = 288K$$

$$Steam\ Load = 100\frac{kg}{hr}$$

$$Steam\ Temperature = 110C = 383K$$

In order to calculate the heat load to produce the steam, the parameters of the water will be needed. Namely, the specific heat will be used to calculate the needed heat to raise the temperature of the water, and the latent heat of vaporization will be used to find the heat needed to complete the phase change from water to steam. These are found from a thermodynamics textbook.

$$Specific\ heat\ (C_p) = 4.22\frac{kJ}{kg}$$

$$Latent\ Heat\ of\ Vaporization\ (h_{fg}) = 2257\frac{kJ}{kg}$$

$$Boiling\ Point\ of\ Water\ (T_b) = 100C = 373K$$

With the necessary information sorted, the heat load can be calculated. Finding the heat load over a phase change in a substance must be done in 2 parts. First, the heat needed to raise the temperature of the water from the initial temperature to phase-change temperature as well as to heat the steam from phase-change temperature to final temperature must be found using equations 1 & 2. Second, the heat consumed during the phase-change is found using equation 3. These two heats are then added together to find the total.

$$Q_1 = \dot{m}_w * C_p * (T_b - T_{wi}) \tag{1}$$

$$Q_2 = \dot{m} * C_p * (T_{wo} - T_{wi}) \quad (2)$$

$$Q_3 = \dot{m} * h_{fg} \quad (3)$$

$$Q_{total} = Q_1 + Q_2$$

Performing the calculations and converting to $\frac{kg}{s}$ reveals that $Q_{total} = 73.823\frac{kJ}{s}$ consumed per dryer in operation.

Heat Exchanger Analysis

Using the rate of heat transfer dictated by the working load, as calculated above, the heat exchanger can now be analyzed in full. The goal of this analysis is to find the mass flow rate and outlet temperature of the oil.

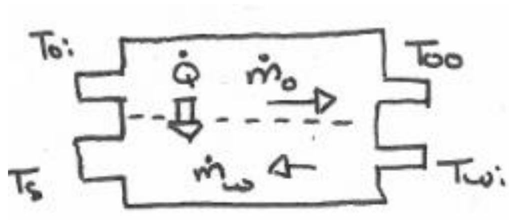


Figure 2: Heat Exchanger Diagram

To begin, variables must first be defined, as shown in figure 2. Given that the heat exchanger in question is a counterflow heat exchanger, the flows of the two fluids are opposite each other. The upper flow, denoted with the subscript "o", represents the hot oil, while the lower flow represents the water/steam. The temperatures are notated accordingly; the mass flow rates denoted with an m, and the heat flow is denoted Q. At this point, the mass flow rate of oil and outlet temperature of oil are unknown.

To find the mass flow rate, which will be used to find its outlet temperature, recall the values of heat transfer drawn by the machines, and convert that value into a total value for shifts one and two:

$$Q_1 = 73.823\frac{kJ}{s} * 2\, machines = 147.646\frac{kJ}{s} \quad (1^{st} shift)$$

$$Q_2 = 73.823\frac{kJ}{s} * 1\, machine = 73.823\frac{kJ}{s} \quad (2^{nd} shift)$$

Using these values for the actual heat transfer, the heat losses in the exchanger must be accounted for. This is done through the use of an effectiveness ratio, provided in the problem as $\varepsilon = 0.8$. In a real engineering application, this would come from the manufacturer of the heat exchanger, where a more in-depth heat transfer analysis as well as experimental data would be used to come to a value.

In general, the effectiveness ratio is defined as

$$\varepsilon = \frac{Q_{actual}}{Q_{maximum}} \quad (4)$$

Where the equation for the maximum possible heat transfer is

$$Q_{maximum} = \dot{m}_w C_{pw}(T_{oi} - T_{wi}) = \dot{m}_o C_{po}(T_{oi} - T_{wi}) \qquad (5a \ \& \ 5b)$$

The above equation comes from the idea that heat transfer always travels down a temperature difference. Applying this idea to the exchanger, the maximum temperature the water can attain is the inlet temperature of the oil, and the minimum temperature the oil can attain is the inlet temperature of the water. Therefore, the maximum temperature difference, as denoted in equations 5a and 5b, is the difference between the inlet oil temperature and inlet water temperature.

Notice also that the respective mass flow rates and specific heats of the two fluids are interchangeable. This is because of the law of conservation of energy. The heat transfer from the oil to the water must be identical provided the same temperature difference (assuming no external heat loss in the ideal case).

Combining equations 4 and 5b, and substituting in the known values of Q_{actual} for first and second shift, the mass flow rates of oil can be found:

$$Q_{maximum} = \frac{Q_{actual}}{\varepsilon} = \dot{m}_o C_{po}(T_{oi} - T_{wi})$$

$$\dot{m}_o = \frac{Q_{actual}}{\varepsilon C_{po}(T_{oi} - T_{wi})} \qquad (6)$$

Plugging the known values for Q_{actual}, ε, and inlet temperatures into equation 6, the mass flow rates are found:

$$\dot{m}_o = 0.587 \frac{kg}{s} \ (first\ shift)$$

$$\dot{m}_o = 0.294 \frac{kg}{s} \ (second\ shift)$$

At this point a good check is to make sure the second shift mass flow rate is one-half the first shift mass flow rate, since the heating load is half, a test which these value pass. Having the values for both mass flow rates will come in handy when determining the sizes for the storage tanks and solar collectors.

To finish off the analysis of the heat exchanger, the final unknown, the outlet temperature of the oil, must be found. By performing an energy balance on the oil, the heat transfer out can be related to the temperature across the oil, and the outlet temperature can be derived. If all of the above steps were done correctly, the outlet temperature should come out the same for both shifts.

$$Q_{actual} = \dot{m}_o C_{po}(T_{oi} - T_{oo})$$

$$T_{oo} = T_{oi} - \frac{Q_{actual}}{\dot{m}_o C_{po}} \tag{7}$$

Substituting into equation 7 the values found above yields oil outlet temperatures as follows:

$$T_{oo} = 314.96K \qquad (first\ shift)$$

$$T_{oo} = 315.14K \qquad (second\ shift)$$

For the purposes of this project, these temperatures are close enough to be considered the same, and from this point forward the oil outlet temperature will be considered 315K.

Storage Tank Sizing

Following the analysis plan laid out earlier, attention is turned to the storage tanks. This is the first place where some engineering decisions will need to be made. These decisions will dictate the necessary parameters for the tank, from which a final size can be determined.

The first engineering decision is how much energy storage capacity, in terms of runtime, would be desired. Using this value, an absolute minimum tank size can be determined, to begin to narrow down the selection. One full week, seven days, has been selected as the minimum necessary storage capacity. In order to find a minimum tank size from this value, the time period of one shift over seven days is first converted into seconds. This value is then multiplied by the rate of heat consumption for first and second shift to find the total heat consumption per week of each shift. Finally, these values are added together to find a total heat consumption by the plant over a week.

$$7\ days * 8\frac{hours}{day} * 3600\frac{seconds}{hour} * 147.646\frac{kJ}{s} = 2.97 * 10^7\frac{kJ}{week} \qquad (1^{st} shift)$$

$$7\ days * 8\frac{hours}{day} * 3600\frac{seconds}{hour} * 73.823\frac{kJ}{s} = 1.488 * 10^7\frac{kJ}{week} \qquad (2^{nd} shift)$$

$$2.97 * 10^7\frac{kJ}{week} + 1.488 * 10^6\frac{kJ}{week} = 4.465 * 10^7\frac{kJ}{week} \qquad (Total)$$

Next, looking at the hot storage tank, the energy stored in the tank at any given time is found through equation 8, with density multiplied by volume in place of mass:

$$E_{tank} = m_{tank} C_{po} T_{tank}$$

$$E_{tank} = \rho_o V_{oil} C_{po} T_{tank} \tag{8}$$

Assuming that the tank is full of oil at the start of the week, the volume of the oil becomes the volume of the tank. By substituting the weekly energy consumption in for tank energy, the volume of the tank can then be solved for:

$$E_{tank} = \rho_o V_{tank} C_{po} T_{oil}$$

$$V_{tank} = \frac{E_{tank}}{\rho_o C_{po} T_{oil}} = \frac{4.465 * 10^7 \frac{kJ}{week}}{2.328 \frac{kJ}{kgK} * 423K * 800 \frac{kg}{m^3}} = 56.7\ m^3$$

Solar Collector Analysis

Now to the most complex portion of this project, the solar collector analysis. The system that will be analyzed in this report is a parallel array of solar collectors, each measuring 4ft wide by 8ft long. This setup has been chosen as the application of the array is in a continuously running process plant. In such an environment, reliability is key. As such, a parallel arrangement provides the highest reliability. The size of each panel has been decided as this is a standard panel size in current applications.

The analysis of the solar collectors will be done with the goal to find a direct relationship between the area of the solar collector and the heat output, as determined by the TMY data. This will allow for the array to be appropriately sized. Since the array of panels is arranged in a parallel form, the analysis will serve to solve for an effective, or total, area, which will then be converted into a number of individual panels.

To begin the analysis of the solar collector, consider the equation for useful heat output:

$$Q_u = A_c[S - U_L(T_m - T_a)] \tag{9}$$

$$\begin{aligned} A_C&: \quad Effective\ Collector\ Area \\ S&: \quad Absorbed\ Solar\ Irradiation \\ U_L&: \quad Coefficient\ of\ Heat\ Loss \\ T_i&: \quad Mean\ Plate\ Temperature \\ T_a&: \quad Ambient\ Temperature \end{aligned}$$

The variable S represents the radiation on the plate collector that has passed through the cover and is absorbed by the plate. S is therefore dependent on properties of the cover, through equation 16:

$$S = G_b(\tau\alpha)_b + G_d(\tau\alpha)_d \tag{16}$$

Where G_b and G_d are the beam and diffuse components of solar radiation landing on the cover of the collector, $(\tau\alpha)$ is the transmittance-absorptance product, a property of the plate collector that will be found later.

The beam and diffuse components of solar radiation will be found from the TMY data used in this project. In the data, the direct normal radiation (DNI) is provided in W/m^2. This represents the amount of beam radiation collected by a flat surface that is normal to the sun's rays at time of data collection. The angle at which the sun is incident to the collector, and the angle that the collector is mounted at, however, are not the same. In this case the radiation must be adjusted based on the incidence angle, or the angle between the normal of the collector and the normal to the beam radiation. Adjustment is done through equation 17, where G_bT is the beam radiation on the tilted surface and θ is the angle of incidence:

$$G_{bT} = (DNI)\cos(\theta) \tag{17}$$

The angle of incidence is calculated from its own equation which is dependent on the latitude of the collection location, the slope of the collector, and the astronomical declination of the earth. The latitude and slope are fixed value, at $Lat. = 45.156°$ and $Collector\ Slope\ (\beta) = 25°$. The declination, being the tilt of the earth's rotational axis, varies throughout the year, and is calculated by equation 18, where n is the day of the year starting at January 1st:

$$\delta = 23.45° \sin\left(360 * \frac{284 + n}{365}\right) \tag{18}$$

Since in this project, the absorbed solar radiation for each day is to be found, equation 12 was entered into the excel sheet of TMY data and calculated for each day of the year.

Using the declination, the cosine of the incidence angle can be found through equation 19.

$$\begin{aligned}\cos(\theta) = \sin(\delta)\sin(\varphi)\cos(\beta) - \sin(\delta)\cos(\varphi)\sin(\beta)\cos(\gamma) +\\ \cos(\delta)\cos(\varphi)\cos(\beta)\cos(\omega) + \cos(\delta)\sin(\varphi)\sin(\beta)\cos(\omega)\end{aligned} \tag{19}$$

Equation 19, it must be noted, assumes that the solar collectors are facing due south, such that the solar azimuth angle (γ) is zero. Such an arrangement is, in fact, a valid choice in Maine. Since Maine is north of the tropic of Cancer ($23.45°N$), at all points throughout the year the sun is never directly overheat, but rather some degree towards the south. This means that a southern facing collector will have a lower angle of incidence with the sun and therefore a higher collection of solar radiation.

In order to calculate the diffuse radiation on the surface, an effective angle of incidence is used, the equation for which is given below:

$$\theta_{effective} = 59.7° - .1388\beta + .001497\beta^2 = 57.165°$$

Using this angle of incidence, and plugging into equation 11 for every data entry, the beam and diffuse radiation for each hour is found and stored in the datasheet.

Recalling equation 16, the final variable to be determined is the transmittance-absorptance product, or $(\tau\alpha)$. This value, a fixed property of the collector, is dependent on two properties: the transmittance of the glass cover, or how much radiation it allows to pass through, and the absorptance of the plate, or how much radiation is absorbed instead of reflected out. The trick to finding the transmittance-absorptance product is that these values are not the same across all wavelengths of light. Therefore, a more complex calculation must be done.

In order to find the transmittance-absorptance product of the collector, the following integral must be evaluated:

$$(\tau\alpha) = 1.01 * \frac{\int_0^\infty \tau_\lambda \alpha_\lambda I_{\lambda,i}}{\int_0^\infty I_{\lambda,i}} \tag{20}$$

Equation 20 is the integration of the product of transmittance and absorptance across the radiation spectrum, divided by the radiation over the spectrum to find the overall product. That is then multiplied by 1.01 to approximate the transmittance-absorptance product (not to be confused with the product of transmittance and absorptance) for the collector.

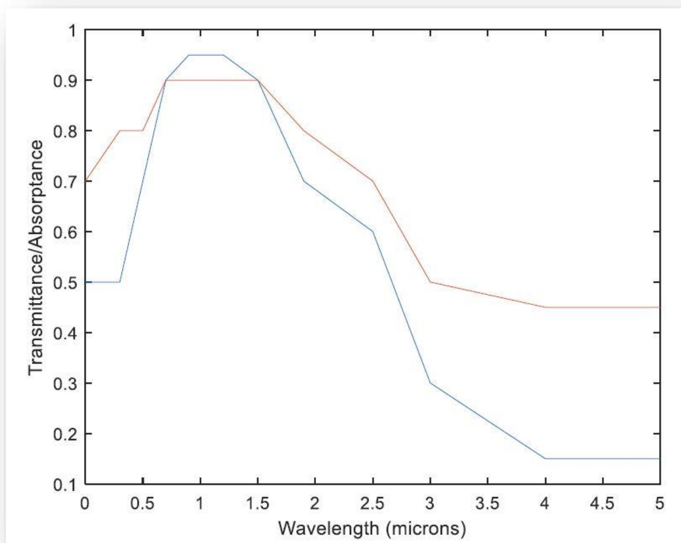


Figure 3: Plot of Transmittance (cover) and Absorptance (plate)

The integral can be performed using a summation notation over the wavelengths specified in the transmittance-absorptance plot provided in the project, seen here as figure 3. To do so, equation 20 is re-written as:

$$(\tau\alpha) = \sum_{N=1}^{N} \tau_\lambda \alpha_\lambda \left(\frac{I_{\lambda,i}}{\sum I_{\lambda,i}}\right) \tag{21}$$

Next, the quantity of incident radiation divided by total incident radiation, $\left(\frac{I_{\lambda,i}}{\sum I_{\lambda,i}}\right)$, is substituted for the solar fraction, as found in table 3.6.1a in the course textbook. Here the values are tabulated by λT. In this case T is the temperature of the sun, or 5777K. Using figure 3.6.1a (right) as well as the transmittance-absorptance plot provided in the project (figure 4), the summation is evaluated for segments divided by the inflection points of the graph. This makes reading the values easy and allows for more accuracy in measurement. Evaluating the summation reveals that $(\tau\alpha)_b = 0.6645$.

Table 3.6.1a Fraction of Blackbody Radiant Energy between Zero and λT for Even Increments of λT

λT, μm K	$f_{0-\lambda T}$	λT, μm K	$f_{0-\lambda T}$	λT, μm K	$f_{0-\lambda T}$
1,000	0.0003	4,500	0.5643	8,000	0.8562
1,100	0.0009	4,600	0.5793	8,100	0.8601
1,200	0.0021	4,700	0.5937	8,200	0.8639
1,300	0.0043	4,800	0.6075	8,300	0.8676
1,400	0.0077	4,900	0.6209	8,400	0.8711
1,500	0.0128	5,000	0.6337	8,500	0.8745
1,600	0.0197	5,100	0.6461	8,600	0.8778
1,700	0.0285	5,200	0.6579	8,700	0.8810
1,800	0.0393	5,300	0.6693	8,800	0.8841
1,900	0.0521	5,400	0.6803	8,900	0.8871
2,000	0.0667	5,500	0.6909	9,000	0.8899
2,100	0.0830	5,600	0.7010	9,100	0.8927
2,200	0.1009	5,700	0.7107	9,200	0.8954
2,300	0.1200	5,800	0.7201	9,300	0.8980
2,400	0.1402	5,900	0.7291	9,400	0.9005
2,500	0.1613	6,000	0.7378	9,500	0.9030
2,600	0.1831	6,100	0.7461	9,600	0.9054
2,700	0.2053	6,200	0.7541	9,700	0.9076
2,800	0.2279	6,300	0.7618	9,800	0.9099
2,900	0.2506	6,400	0.7692	9,900	0.9120
3,000	0.2732	6,500	0.7763	10,000	0.9141
3,100	0.2958	6,600	0.7831	11,000	0.9318
3,200	0.3181	6,700	0.7897	12,000	0.9450
3,300	0.3401	6,800	0.7961	13,000	0.9550
3,400	0.3617	6,900	0.8022	14,000	0.9628
3,500	0.3829	7,000	0.8080	15,000	0.9689
3,600	0.4036	7,100	0.8137	16,000	0.9737
3,700	0.4238	7,200	0.8191	17,000	0.9776
3,800	0.4434	7,300	0.8244	18,000	0.9807
3,900	0.4624	7,400	0.8295	19,000	0.9833
4,000	0.4809	7,500	0.8343	20,000	0.9855
4,100	0.4987	7,600	0.8390	30,000	0.9952
4,200	0.5160	7,700	0.8436	40,000	0.9978
4,300	0.5327	7,800	0.8479	50,000	0.9988
4,400	0.5488	7,900	0.8521	∞	1.

Figure 4: Table 3.6.1a

All that remains is to find the product for diffuse radiation. To do this, graph 5.6.1 from the text is used, which shows the relationship of the product for varying angles of incidence to that of normal radiation. Substituting the effective angle of incidence for diffuse radiation found earlier reveals that $(\tau\alpha)_d = 0.8$.

With the transmittance-absorptance products found, then the value of absorbed solar radiation can be found, on a per-unit-area basis, for each time interval in the dataset using equation 16.

The next variable to tackle is the overall loss coefficient, U_L.This coefficient is used to account for heat losses through various means in the collector. In order to find it, a resistance network for the solar collector must be established:

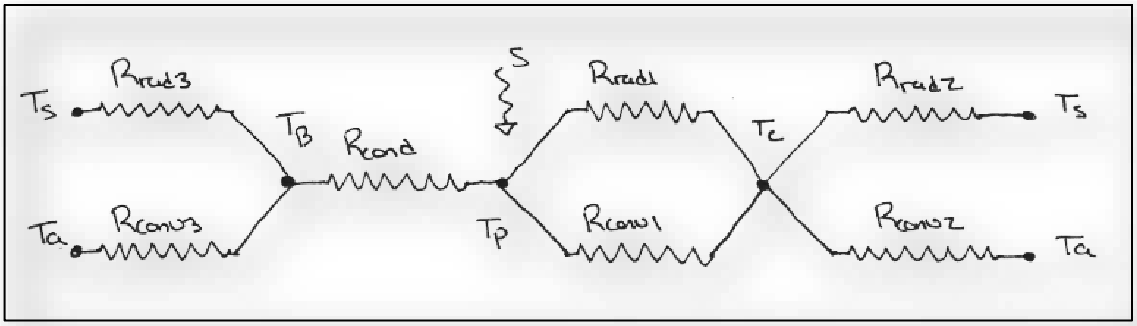


Figure 5: Resistance Network

Using the resistance network as a guide, the individual resistances will be found such that the overall heat loss coefficient may be found using equation 16:

$$U_L = \frac{1}{R_{tot}} \tag{10}$$

The first resistances to be found will be those of the convection losses due to wind over the collector. The process for the front and back of the collector are identical, and so for simplicity the notation in this section will be for the front only.

For the top convective resistance, the heat loss coefficient is as follows:

$$h_{c,2} = \frac{Nu_2 k_{air}}{L_{char}} \tag{11.1}$$

Nu represents the Nusselt number, k the thermal conductivity of the air, and L_{char} is the characteristic length of the collector. For a 4ft by 9ft rectangular collector, as is the case here, the characteristic length is 5.33ft, or 1.63 meters, according to its definition:

$$L_{char} = \frac{4A_p}{perimeter} = \frac{128ft^2}{24ft} = 5.33ft = 1.63m$$

The thermal conductivity of air can be found online to be $k_{air} = 0.1\frac{w}{mk}$. Finally, the Nusselt number must be calculated using the relationship.

$$Nu = 0.83Re^{\frac{1}{2}}\mathrm{Pr}^{\frac{1}{3}} \tag{11.2}$$

In equation 17.1, the Reynolds number (Re) and Prandtl number (Pr) are used. The Reynolds number, a dimensionless quantity used in fluids to describe flow characteristics, is found using equation 11.3, where the Prandtl number, the ratio of momentum and thermal diffusivity, if found using equation 11.4.

$$Re = \frac{Vel * L_{char}}{\gamma} \tag{11.3}$$

$$\Pr = \frac{\gamma}{\alpha} = \frac{1.47 * 10^{-5}}{2.01 * 10^{-5}} = 0.73 \tag{11.4}$$

Since the Reynolds number is dependent on air velocity, which varies throughout the year, it is plugged into excel to be determined for each day using equation 17.2. The Nusselt number is then also calculated for each day of the year. Finally, for each day, the coefficient of heat loss for convection to the atmosphere on the bottom of the collector is calculated in excel.

The next resistance to be found will be found is the conduction from the plate to the bottom of the collector. Since this resistance is the only passage of heat through the back, its calculation is simple:

$$R_{cond} = \frac{\Delta x}{k_{ins}} = \frac{0.12\, m}{.35\, \frac{w}{mk}} = 0.343\, \frac{m^2 K}{w}$$

Before continuing, there is an issue to be addressed. When finding the convective and radiative factors to and from the cover or from the back, the intermediate temperatures between ambient and the plate temperature must be known. However, these temperatures cannot be found without knowing the heat resistances. Therefore, an iterative solution will be necessary. To do so, the following procedure will be employed:

1. Guess the cover/back temperature; somewhere between fluid and ambient.
2. Use this temperature to find the heat transfer resistances and the overall heat loss coefficients for the top/back of the collector.
3. Determine the flow of heat through the top/bottom of the collector using the equation: $q = U_{L,T}\left(T_p - T_a\right)$
4. Solve for the cover/back temperature using the equation: $q = \frac{T_p - T_c}{R}$, where R is the resistance from the cover to ambient, or from the back to ambient.
5. Compare temperatures, and repeat as necessary

Since some of the heat transfer coefficients are dependent on the air velocity over the collector, which varies with time, this iterative process will be done for each datapoint in the dataset to find an accurate heat loss coefficient for all time intervals included.

To perform the iteration on each datapoint, the data is called into MatLab where loops are used to perform the calculations. That same code is then used to calculate the overall useful heat per collector, as will be covered later.

Continuing the calculation of the heat loss coefficients, the radiation loss from the back of the collector will be found using equation 12.1.

$$h_{r,3} = \frac{\sigma(T_B + T_s)(T_B^2 + T_s^2)(T_B - T_s)\varepsilon_c}{T_B - T_a} \quad (12.1)$$

In equation 12.1, T_s represents the sky temperature, which the heat is radiating to. This if found as follows:

$$T_s = T_a\left[.711 + .0056T_{dp}^2 + .013\cos(15t)\right]^{\frac{1}{4}} \quad (12.2)$$

Equation 12.2 is a conversion from the ambient temperature to an approximate sky temperature using the dew point temperature, which is given in the data. The variable “t” represents the number of hours after midnight of the datapoint.

The heat loss coefficient for radiation from the cover to the sky, $h_{r,2}$, matches the form of equation 12.1, with the exception that T_B is replaced with T_c, the cover temperature.

The radiation losses between the plate and the cover are found with an equation similar to 12.1.

$$h_{r,1} = \frac{\sigma\left(T_p + T_c\right)\left(T_p^2 + T_C^2\right)}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_c} - 1} \quad (12.3)$$

In equation 12.3, the mean plate temperature is called on. Traditionally, this temperature would be found using an iterative method like the one above for cover temperature. In this project, since the thermal conductivity of the plate is very high, the assumption that the mean plate temperature is the same as the mean fluid temperature is valid. The mean fluid temperature in this case, is given by the prompt to be the average of the cold and hot storage temperatures. This works out to be 369K

Additionally, ε_p and ε_c represent the emittance of the plate and collector, respectively. For the insulation on the back of the collector, emittance is provided in the problem as 0.7. The emittance of the glass and plate, however, is not provided and must therefore be calculated.

To find the emittance for the cover, the transmittance is first found similarly to how the transmittance-absorptance product, which is found below.

$$\tau = \sum_{N=1}^{N} \tau_\lambda \Delta f d\lambda \quad (13)$$

Where Δf is the solar fraction, as previously defined. Using this method, transmittance is found to be 0.775. From there, transmittance can be used to find absorptance, which in turn will solve for emittance.

$$\varepsilon = \alpha = 1 - \tau \tag{14}$$

Using this method, emittance of the cover is found to be .225.

Solving the above sum, substituting absorptance for transmittance, the emittance for the plate is found to be 0.835

The convective loss from the plate to the cover is found similarly to the convection due to wind, however in this case it is natural convection. That calls for a different Nusselt number formula as well as the Rayleigh number:

$$h_{c,2} = \frac{Nuk_{air}}{L} \tag{15.1}$$

$$Nu = 1 + 1.44\left[1 - \frac{1708\sin(1.8\beta)^{1.6}}{Ra\cos(\beta)}\right]\left[1 - \frac{1708}{Ra\cos(\beta)}\right]^{+} + \left[\left(\frac{Ra\cos(\beta)}{5830}\right)^{\frac{1}{3}} - 1\right]^{+} \tag{15.2}$$

$$Ra = \frac{g\beta'\left(T_p - T_c\right)L^3}{\gamma\alpha} \tag{15.3}$$

Note: in equation 15.2, the "+" exponent indicates that only the positive values will be used. If the expression in brackets works out to be negative, then the expression is set equal to zero.

The final consideration for the solar collector is the critical radiation level. Simply, the level of radiation, below which the fluid will lose heat between the inlet and outlet. The critical radiation factor is found using equation 22:

$$G_{Tc} = \frac{U_L(T_i - T_a)}{(\tau\alpha)} \tag{22}$$

Since the overall loss coefficient is different for different times, critical radiation level is calculated for each datapoint. If the radiation is below this level, then the absorbed solar radiation is set to zero. This check is done by comparing the absorbed solar radiation, S, with G_{Tc} at all points.

Programming Process

The processes described above in this project depend heavily on programming in both MATLAB and Microsoft Excel. The initial calculations of beam and diffuse radiation, angle calculations, and calculations for the absorbed solar radiation were done in excel from the data download. From there, the table was imported into MATLAB as an array, and the heat loss analysis on the collector was performed. This includes the iterative solution for the cover and back temperatures, as well as iterating over all the days of the year. The result is a matrix that includes, for one year in 1-hour intervals, data for the following:

- Heat demand from the dryers
- Absorbed solar radiation per collector panel
- Overall heat loss coefficient for every point

Sizing the Collector System

For the engineering decisions, the first and simplest one will be how many collectors will be needed. To answer that, the total load for the year is summed up and divided by the total useful heat extracted per collector panel. Performing the calculation reveals that given the collector analyzed above, 301 panels would be needed. That is an effective area of 895 m^2.

Sizing the storage tanks for the system is a bit trickier. Based on the calculations earlier, the tank must be at least 57 cubic meters. As per the problem statement, this tank will be oversized by 50% for a factor of safety to come to 85.5 cubic meters. This number comes from the decision that at 100% tank capacity, the dryers should have a week of run-time with no solar input. However, Maine presents an additional challenge. Being located at 45 degrees latitude, the mill sees significant variation in sunny hours throughout the year. In the winter months there is much less radiation to collect than in the summer. In order to determine if the tank is sized appropriately, the tank level must be graphed over the entire year. Based on the tank beginning at maximum capacity, a properly sized tank will never run out of oil. Using MatLab and analysis methods described below, the level of the tank over the

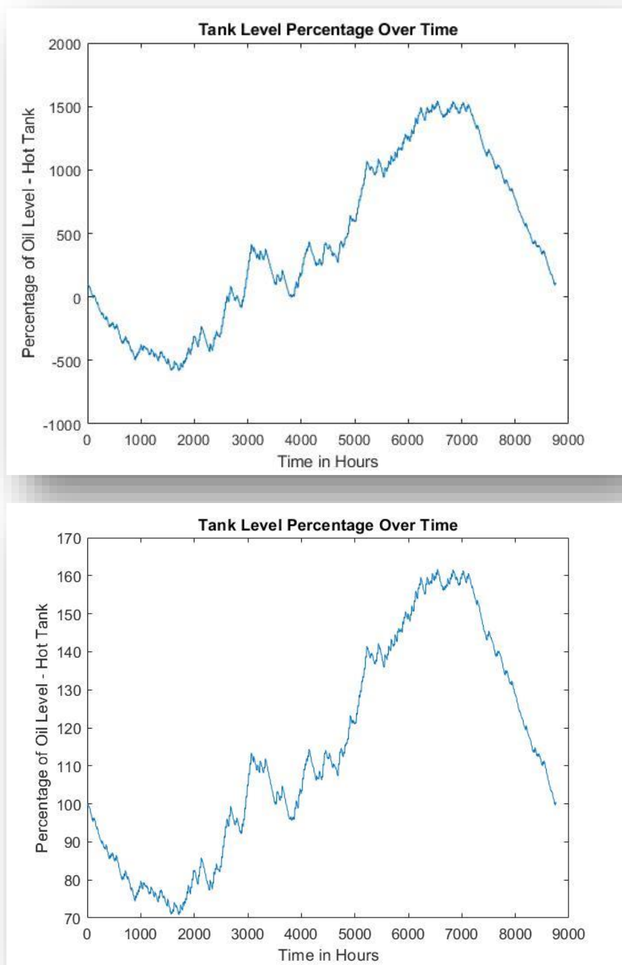


Figure 6: Tank Level (%) Yearly at (top) 85.5 cubic meters, and (bottom) 2000 Cubic Meters

entire year was plotted as a percentage. The results are shown in figure 6 (top). The result is that the tank level varies from -500% to just over 1500% of its rated capacity. Of primary concern is the negative, however, as this represents running out of hot oil. If the storage capacity is then increased to 2000 cubic meters, simulations predict that the level in the tank will remain sufficient throughout the year. In fact, for much of the year in the summer there is a surplus of energy, as shown in figure 6 (bottom).

The surplus of energy which remains in the enlarged system must be addressed. For a mill located in a more consistently sunny area, the solar array would simply be downsized, but since the solar variation throughout the year in Maine is high, there will always be significant surpluses during the summer months. As a solution to this problem, the system could incorporate another heat exchanger and steam generator. This one would feed into an electric turbine to generate power. Currently, there are two steam turbines installed at the mill, and most mills have a similar setup due to the high amount of steam produced in the pulp making process. Another option would be to incorporate a second heat exchanger into the recovery process, where spent chemicals are boiled in a large boiler to be reused.

Regardless of specific use, pulp and paper mills have near endless uses for pure thermal energy, such that siphoning off the surplus in the summer would not pose a significant challenge, and would add to the economic benefits of a solar investment.

Weekly Analysis

To perform a more detailed analysis, one week is selected from the year to analyze. The first week of February has been selected. Additionally, the analysis will be based on the larger, optimized tank of 2000 cubic meters.

In order to plot understand how the level of the tank changes over time, the change of mass in the tank over time is first derived. Based on an initial tank level, the initial mass if found and through a mass balance of the tank, equation 23 is derived:

$$M_{HT} = M_i + \int_0^t \dot{m}_{collector} - \dot{m}_{load} dt \quad (23)$$

Here, M_{HT} represents the mass present in the hot storage tank, M_i the initial mass present, $\dot{m}$ represents the mass flow rates, and t represents time since the initial mass value was taken. The mass flow rates for the load were found in the analysis of the heat exchanger, but the mass flow rate for the collector will need to be derived.

Considering an energy balance on the entire solar collector, it is known that the rate of heat flow into the fluid can be found thusly:

$$\dot{Q} = \dot{m} C_p \Delta T$$

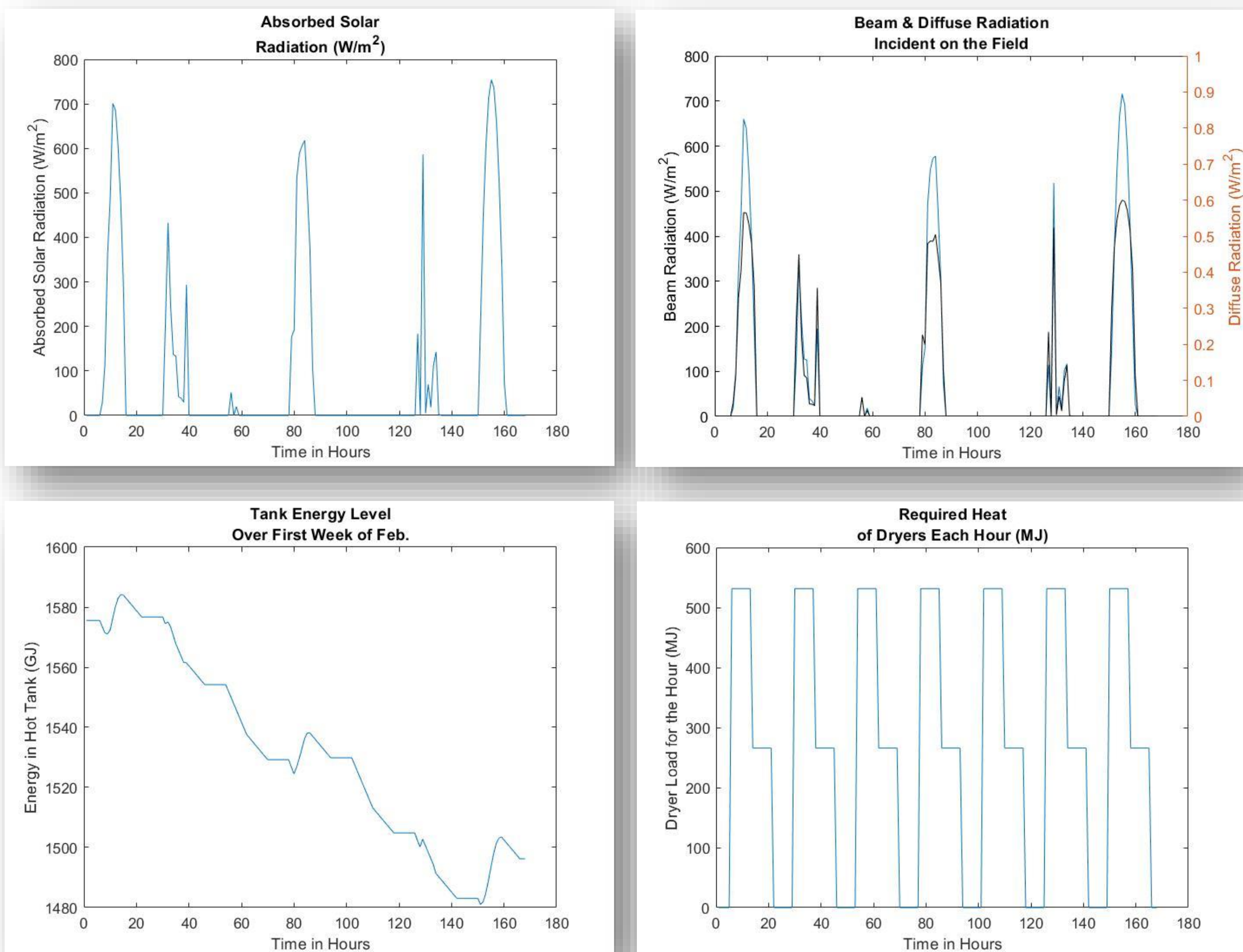
This equation is to be applied to the heat coming in from the collector, as the mass flow rates from the dryers were calculated earlier. Rearranging the equation to solve for mass flow rate, provides an expression for mass flow rate that can be applied to the data, as demonstrated in equation 24.

$$\dot{m} = \frac{\dot{Q}}{C_p \Delta T} \tag{24}$$

To find the energy stored, the percentage total mass in the tank is converted to energy using the temperature and specific heat:

$$E_{tank} = \%Full * (m_{tank} C_{p,oil} T_{tank}) \tag{25}$$

Using Matlab and the data collected for incident solar, plots can be made for the week:



Discussion & Conclusions

In analyzing a solar-thermal collection system for steam generation in pulp drying, solely for the purposes of running the dryers, it would not be a particularly efficient system given the location. There is too much yearly variation in solar energy available to allow for a system with minimal surpluses or shortfalls. Any system, therefore, installed for just one purpose will either be insufficiently powered, requiring supplementation from other sources, or will be oversized drastically over the summer months, needing to be throttled in order to not overload the system.

To solve the surplus problem, in a parallel arrangement of panels, some panels could be shut off, but this would leave them idle for much of the year, and a wasted investment. A solar-thermal collection system, similar to the one proposed in this project, would be a viable option if more than one sink for the thermal energy was connected. This way, the surplus energy in the summer would be used elsewhere in the plant increasing the return on investment and reducing operating costs.

For this mill, a 2000 cubic meter, or 2,000,000 liter, storage capacity. This would provide a maximum energy storage capacity of 3,724,000 kJ, enough to run the plant for 35 weeks, which is nearly half the year. While this is far and above what was needed initially, at 1 week, this much capacity is necessary due to the large fluctuations throughout the year, as discussed above. While 2000 cubic meters seems like a very large storage vessel, that volume is equal to the average hot air balloon size. Considering that the recovery boiler of the mill in this project is 14 stories tall, such a vessel is not implausible.

Connected to a field of 301 panels, totaling 895 m^2, the system would be sufficient to be able to run the dryers entirely on solar energy. While there would be significant surpluses, as discussed above, these are far superior to an undersized system where supplemental heating would be necessary, adding complexity to the system and more potential points of inconsistency in drying ability.